

Output characteristics of Stirling thermoacoustic engine

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Abstract

A thermoacoustic engine (TE), which converts thermal energy into acoustic power by the thermoacoustic effect, shows several advantages due to the absence of moving parts, such as high reliability and long lifetime associated with reduced manufacturing costs. Power output and efficiency are important criteria of the performance of a TE. In order to increase the acoustic power output and thermal efficiency of a Stirling TE, the acoustic power distribution in the engine is studied with the variable load method. It is found that the thermal efficiency is independent of the output locations along the engine under the same acoustic power output. Furthermore, when the pressure ratio is kept constant at one location along the TE, it is beneficial to increasing the thermal efficiency by exporting more acoustic power. With nitrogen of 2.5 MPa as working gas and the pressure ratio at the compliance of 1.20 in the experiments, the acoustic power is measured at the compliance and the resonator simultaneously. The maximum power output, thermal efficiency and exergy efficiency reach 390.0 W, 11.2% and 16.0%, which are increased by 51.4%, 24.4% and 19.4%, respectively, compared to those with a single R–C load with 750 ml reservoir at the compliance. This research will be instructive for increasing the efficiency and making full use of the acoustic energy of a TE.

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1. Introduction

Wide attention has been focused on the TE for its potentially high efficiency, simple configuration, reliability and environmentally friendly working gas. A TE is an ideal driver for pulse tube cryocoolers (PTCs) and other thermoacoustic refrigerators (TRs). Power output and efficiency are two important parameters of a TE. Up to now, most attention has been focused only on the acoustic power output [1,2], and the acoustic power output is measured with one load each time in most literatures, which is called the single output method in this paper. In fact, the interior acoustic field of a TE is the basis for power output. There is a close relationship between the acoustic power distribution inside the engine and the acoustic power output, which should be paid more attention.

A Stirling TE possesses a torus and a branch resonator, which is more complicated than the standing wave TE [3,4]. It can obtain higher efficiency for its reversible thermoacoustic conversion processes. The engine converts thermal energy into acoustic power in the thermoacoustic core, which comprises an ambient heat exchanger, a regenerator and a hot heat exchanger. Then, the acoustic power is transferred to the tee where it is divided into two parts. One part goes to the resonance tubes, and the other part returns along the feedback tubes. Theoretically, acoustic power can be exported simultaneously from different locations along either the resonance tubes or the feedback tubes. In this paper, the effects of R–C loads at different locations on the acoustic power output and acoustic power distribution inside the engine are studied. Then, we propose to set two R–C loads along the engine simultaneously to increase the acoustic power output and the thermal efficiency, which is called the double output method, to be distinguished from the single output method [5]. As is verified

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by the computational and experimental results, the power output and the efficiency of the engine with double outputs are significantly increased.

2. Computational results

2.1. Introduction of the engine and computation method

Fig. 1 shows the schematic diagram of the Stirling TE. Detailed configuration parameters and performance have been introduced in Refs. [4,6]. Acoustic power is exported from two locations. Location 1 is at the compliance in the torus, and Location 2 is at the start point of the resonator. The R–C variable load method is adopted for acoustic power computation and measurement [2,7]. Fig. 2 shows the schematic diagram of the R–C load, which consists of a reservoir, a needle valve and two pressure sensors. In the computations and experiments, the volumes of the two reservoirs are 500 ml and 750 ml, respectively.

Since the dimensions of the R–C load are much smaller than the acoustic waveguide of the TE, it can be modeled as two lumped impedances, an acoustic resistance and an acoustic compliance. The compliance impedance can be calculated from the reservoir volume V and working condition. If the dynamic pressure p_T is known, the volumetric flow U through the needle valve can be determined as

$$U = \frac{i\omega V p_T}{\gamma p_m} \quad (1)$$

where i is the imaginary unit, γ is the heat capacity ratio, ω is the angular frequency and p_m is the mean pressure. The acoustic power dissipated through the needle valve is defined as

$$E_{\text{load}} = \frac{\omega V |p_T| |p_E| \sin \theta}{2\gamma p_m} \quad (2)$$

where $|p_E|$ and $|p_T|$ are the pressure amplitudes before and after the valve and θ is the phase difference between p_E and p_T . The resistance of the valve can be calculated by

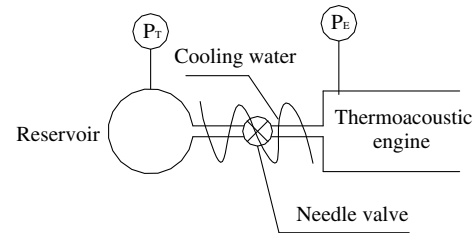


Fig. 2. Schematic diagram of acoustic power measurement. p_E and p_T show the locations of the two pressure sensors.

$$R = \frac{p_E - p_T}{U} \quad (3)$$

Apparently, the acoustic power dissipated through the valve is the acoustic power output of the TE.

The computation is performed by using DeltaE (design environment for low amplitude thermoacoustic engines), which was developed by W.C. Ward and G.W. Swift. Based on linear thermoacoustics theory, DeltaE includes modules for heat exchangers, regenerators, ducts, pistons etc., so that a complete Stirling TE can be modeled [8,9]. We model the self made TE (shown in Fig. 1) as 64 modules connecting in turn, including the torus and resonance tubes. Viscous dissipation and nonlinear dissipative effects, such as dissipation associated with bends and tapered ducts, are considered. Nitrogen of 2.5 MPa is used as the working gas.

2.2. Effect of output location with single output method

The engine shown in Fig. 1 is a 1/4 wavelength Stirling TE. Its pressure ratio becomes maximal just above the main cooler, decreases gradually anticlockwise along the torus and reaches the pressure node before the tapered resonance tube [10]. So, the acoustic power output should be different at different locations due to the acoustic field distribution. In the computation, the pressure ratio at the compliance is kept at 1.20.

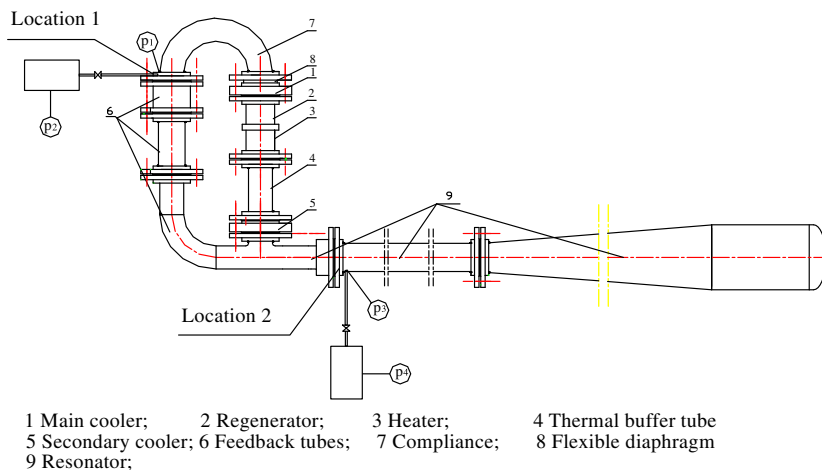


Fig. 1. Schematic diagram of the thermoacoustic engine.

Fig. 3 shows the acoustic power output dependence on resistance with the single output method at different locations. All the computational output curves have similar trends. The curves peak when the resistance is in the range of $3.0\text{--}5.0 \times 10^7 \text{ N s/m}^5$. The compliance impedances of the 500 ml and 750 ml reservoirs are $4.55 \times 10^7 \text{ N s/m}^5$ and $3.03 \times 10^7 \text{ N s/m}^5$, respectively, which indicates that the acoustic power output reaches its maximum when the resistance and the compliance impedance of the R–C load are equal. The computational results also show that the engine can export more acoustic power with a reservoir of larger volume and a valve of larger flow coefficient (i.e. smaller resistance). The acoustic power outputs at the compliance are much larger than those at the start point of the resonance tubes, which is due to the acoustic field distribution. When the resistance of the R–C load is larger than $1.0 \times 10^8 \text{ N s/m}^5$, it has a dominant influence on the output characteristics compared to that of the compliance impedance. So, the output curves with different reservoirs at the same location are almost superposed.

Fig. 4 shows the acoustic power distribution in the TE with the R–C load with 500 ml reservoir at the start point of the resonance tubes. When the resistance is less than $1.0 \times 10^5 \text{ N s/m}^5$ or larger than $1.0 \times 10^{10} \text{ N s/m}^5$, about 330 W acoustic power is transferred to the resonator and dissipated to maintain the interior acoustic field. About 500 W acoustic power is generated by the thermoacoustic core, while the acoustic power returned to the ambient end of the regenerator is about 875 W, which indicates that more acoustic power can be exported from the torus. Fig. 5 shows the acoustic power distribution with the R–C load with 750 ml reservoir at the compliance in the torus. More acoustic power is returned at the tee, and more acoustic power is exported as expected. The more important result is that the TE generates much more power in the thermoacoustic core when the power output curve rises. As is

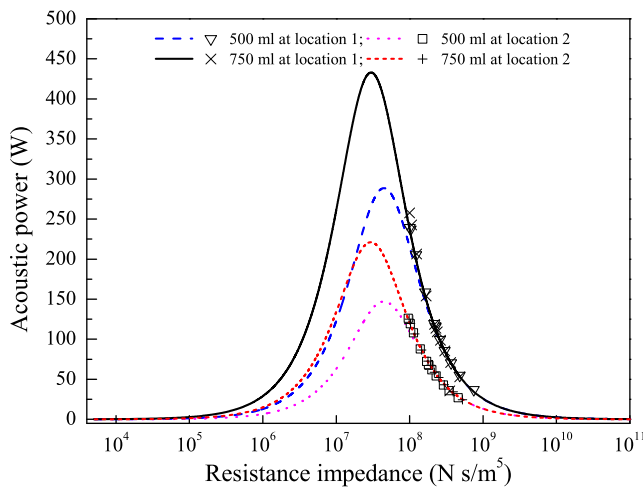


Fig. 3. Comparison of single power outputs at different locations. Solid, dash, short dash and dot curves are computational results. Scattered points are experimental results. 500 ml and 750 ml denote the related reservoirs of 500 ml and 750 ml for conciseness.

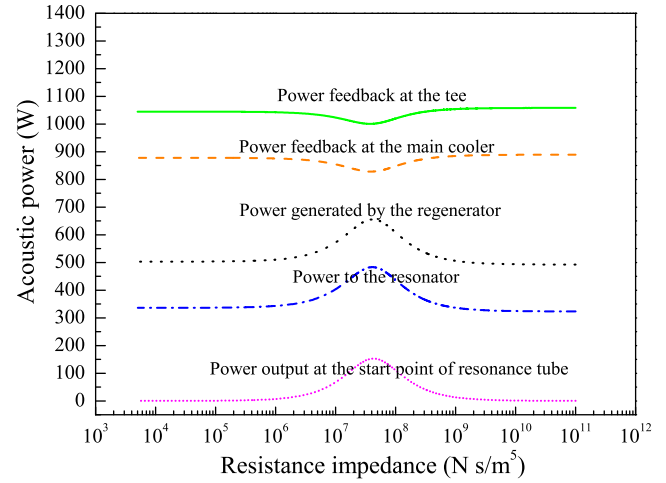


Fig. 4. Acoustic power distribution in the engine with the R–C load with 500 ml reservoir at the start point of the resonance tubes.

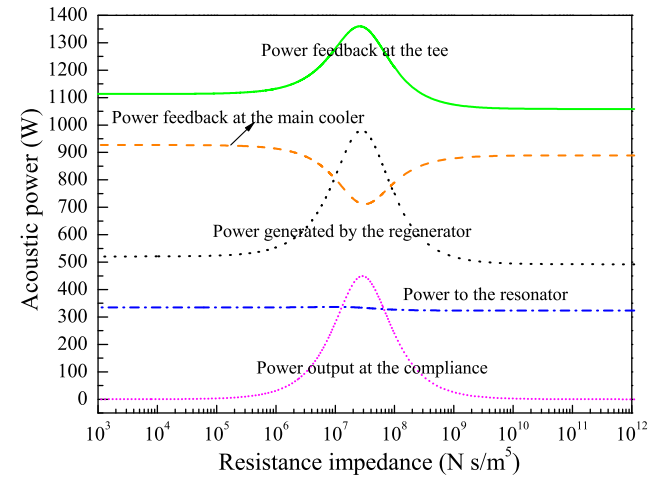


Fig. 5. Acoustic power distribution in the engine with the R–C load with 750 ml reservoir at the compliance in the torus.

shown, when the power output peaks, the feedback acoustic power at the main cooler reaches its minimum, while the power generated in the core peaks. Comparison with Fig. 4 indicates that the regenerator works efficiently here. It is also shown that the acoustic power transferred to the resonator is kept nearly constant when the variable R–C load is set at the compliance.

In order to make clear where acoustic power can be exported more efficiently, the dependences of the thermal efficiency and exergy efficiency on acoustic power output at different output locations are shown in Fig. 6. The definitions of thermal efficiency and exergy efficiency of a TE are

$$\eta = \frac{W}{Q} \quad (4)$$

$$\eta_e = \frac{W}{Q \left(1 - \frac{T_0}{T_H} \right)} \quad (5)$$

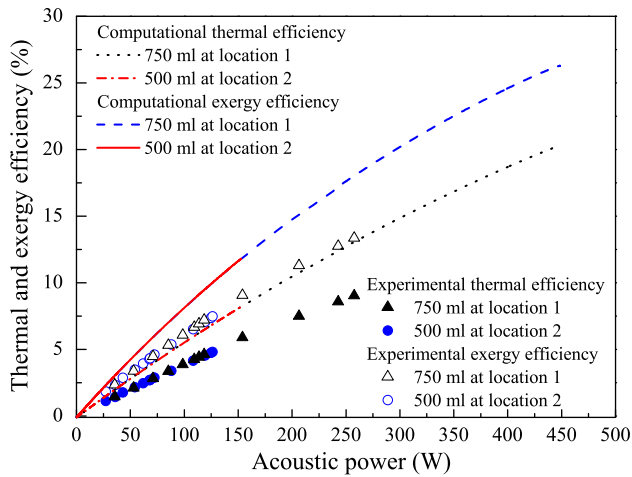


Fig. 6. Thermal and exergy efficiency dependence on acoustic power output. Solid, dash, dot and dash dot curves are computational results. Scattered points are experimental results. 500 ml and 750 ml denote the related reservoirs of 500 ml and 750 ml for conciseness.

where W is the net acoustic power output to the R–C load, Q is the heating power, T_0 is ambient temperature and T_H is the wall temperature of the heater. The acoustic energy stored in the interior acoustic field is dissipated by the TE itself. When calculating the thermal and exergy efficiencies, we only take the acoustic power exported by the R–C loads into account, which is different from Ref. [11]. If there is no net acoustic power output, the efficiency of the TE is zero. Two conclusions are drawn from the comparison of the curves. One is that the thermal efficiency and exergy efficiency are independent of power output location under the same acoustic power output. Although the acoustic power outputs at different locations are different, even with the same R–C load (shown in Fig. 3), the thermoacoustic conversion efficiencies are at the same level so long as the acoustic power outputs are equal. The other is that the thermal efficiency and exergy efficiency increase with acoustic power output. For the pressure ratio at the compliance of the TE remaining at 1.20, the power dissipated in the whole engine changes little during the whole operation, which means that more acoustic power output leads to lower average dissipation. Comparing Fig. 6 with Fig. 3, we can find that much more acoustic power output and higher efficiency can be obtained by exporting acoustic power at the compliance.

2.3. Performance with double output method

Inspired by the above results, we then use the so called double output method to export more acoustic power with two R–C loads simultaneously (as shown in Fig. 1). In order to obtain the trends of acoustic power output with two loads, the valve at the resonator (Location 2 in Fig. 1) is fully opened first (by decreasing the resistance in the computation), and the valve at the compliance

(Location 1 in Fig. 1) is gradually opened. In the computation, the pressure ratio at the compliance is also kept at 1.20.

Fig. 7 shows the comparison of power outputs between the single and double output methods. All the computational curves peak at particular resistances. On the right side of the peak, the total acoustic power output increases with the opening of the valve at the compliance. The total acoustic power with the double output method is approximately the sum of the powers with the single output method, which implies that the outputs at different locations have little influence on each other. During the variation of the resistance at Location 1, the power output with the 500 ml reservoir at Location 2 varies little. With the 750 ml reservoir at Location 1 and the 500 ml reservoir at Location 2, the total acoustic power output reaches its maximum. When the resistance and compliance impedances are nearly equal, the maximal acoustic power output is about 550 W, while the maximal power output is 430 W with the single output method.

Fig. 8 shows the comparison of the thermal efficiencies between the single and double output methods. Keeping the resistance the same, the thermal efficiency with the double output method is much higher than that with the single output method. The highest computational thermal efficiency reaches 23.5% when the resistance is about $2.9 \times 10^7 \text{ N s/m}^5$, which is increased by 18.7% more than the single output thermal efficiency of 19.8%. Comparison of the exergy efficiencies is shown in Fig. 9. The maximal exergy efficiency with the single output method is 25.7%, while it reaches about 30.0% with the double output method, increased by 15.6%.

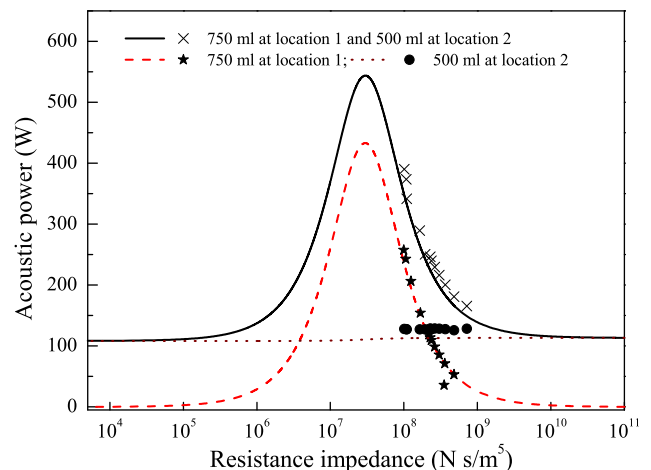


Fig. 7. Comparison of power outputs between single and double output methods. Solid, dash and dot curves are computational results. Scattered points are experimental results. 500 ml and 750 ml denote the related reservoirs of 500 ml and 750 ml for conciseness. The powers denoted by 500 ml and 750 ml show the individually exported acoustic powers, respectively, with double output method.

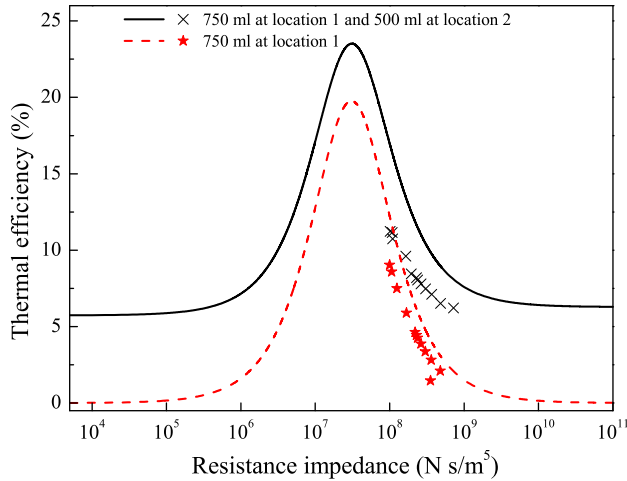


Fig. 8. Comparison of thermal efficiencies between single and double output methods. Solid and dash curves are computational results. Scattered points are experimental results. 500 ml and 750 ml denote the related reservoirs of 500 ml and 750 ml for conciseness.

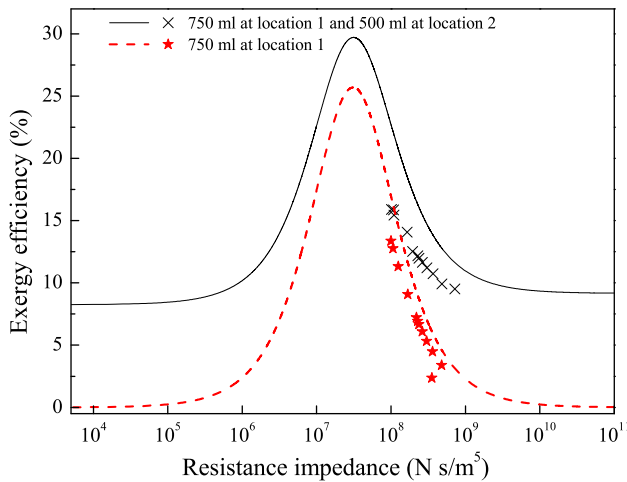


Fig. 9. Comparison of exergy efficiencies between single and double output methods. Solid and dash curves are computational results. Scattered points are experimental results. 500 ml and 750 ml denote the related reservoirs of 500 ml and 750 ml for conciseness.

3. Experimental verification

In order to verify the simulation, a number of experiments were performed. The engine and measurement set up are shown in Figs. 1 and 2. Four calibrated pressure sensors, labeled p_1 , p_2 , p_3 and p_4 , are arranged in the sys-

tem. Two needle valves (Swagelok®) with maximal flow coefficients of 0.73 are applied. In order to keep the gas properties uniform, cooling water is used to cool the tubes and valves. Nitrogen of 2.5 MPa is used as the working gas. The pressure data acquisition system comprises the pressure sensors, data acquisition clip, computer and self developed program based on Labview 6.1 by National Instruments Inc. (NI) [12]. The pressure sensors are of the linear silicon piezoelectric type and were supplied by Infineon Technologies, Germany. The data acquisition clip is NI PC-1200, 12 bits in precision and 10 k S/s in sampling frequency. Calibrated NiCr–NiSi thermocouples, digital multimeter (Prema 5017) together with a digital scanner are used to measure the temperatures. Heating power is adjusted by the charging voltage to the heaters and measured by a digital dynamometer.

As in the computation, the pressure ratio at p_1 is stabilized at 1.20 by adjusting the heating power. The experimental data are scattered in Figs. 3, 6–8 and 9, respectively, and Table 1 shows the maximal values under different conditions.

It is shown in Fig. 3 that the experimental acoustic power output decreases with the increase of resistance impedance since the resistances of the needle valves are on the right side of the peak. In the experimental range, the computational and experimental results are in good agreement. As shown in Table 1, with the 500 ml reservoir at the compliance, the maximal acoustic power output of 239.4 W is obtained, while it is only 126.2 W at the resonator. With the 750 ml reservoir at the compliance, the maximal acoustic power output of 257.6 W is obtained, while it is 125.2 W at the resonator. As shown in Fig. 6, the thermal efficiency and exergy efficiency are also increased greatly by arranging the R–C loads at the compliance by exporting more acoustic power. Resembling the computation curves, the scattered experimental points of thermal efficiency and exergy efficiency are also superposed in their common range, respectively. We did not take serious measures to thermally insulate the heat exchangers, the regenerator and the thermal buffer tube in the experiments, and this heat loss is not included in the computation. Therefore, there is a relatively large deviation between the computational and experimental results.

As shown in Fig. 7 and Table 1, the maximal total acoustic power output with the 750 ml reservoir at Location 1 and the 500 ml reservoir at Location 2 is 390.0 W, which is increased by 51.4% more than that with the single

Table 1
Comparison of experimental results

	Single output				Double output A*
	500 ml reservoir		750 ml reservoir		
	Location 1	Location 2	Location 1	Location 2	
Maximal power (W)	239.4	126.2	257.6	125.2	390.0
Thermal efficiency	8.3%	4.8%	9.0%	4.7%	11.2%
Exergy efficiency	12.3%	7.5%	13.4%	7.3%	16.0%

A* denotes the condition with 750 ml reservoir at Location 1 and 500 ml reservoir at Location 2.

output method. During the adjustment of the valve at location 1, the power output with the 500 ml reservoir at Location 2 varies little.

In Fig. 8, the highest thermal efficiency with the double output method reaches 11.2% when the resistance is about $1.0 \times 10^8 \text{ N s/m}^5$, which is increased by 24.4% more than the single output thermal efficiency of 9.0%. The relatively large deviation between the computational and experimental results is also caused by the poor thermal insulation. It is shown in Fig. 9 and Table 1 that the exergy efficiency is increased remarkably too. The maximal exergy efficiency with the single output method is 13.4%, while it reaches 16.0% with the double output method, increased by 19.4%.

4. Conclusion

Computations and experiments show that the power output locations have not much influence on the thermal efficiency and exergy efficiency under the same power output. Because of the acoustic field distribution, more acoustic power can be exported from the locations with higher pressure ratio (or pressure amplitude) with the same R–C load. When the pressure ratio at one location is kept constant, the acoustic power dissipated inside the engine varies little. So, more acoustic power output leads to higher efficiency.

Compared to mechanical compressors, a TE has a large scale interior acoustic field, which makes it more convenient to export acoustic power from more than one location along its waveguide simultaneously. For simplicity and convenience, we proposed to use two simultaneous outputs to show the performance improvements of a Stirling TE in this paper. More acoustic power and higher efficiency have been obtained. From the viewpoint of applications, it will be convenient and efficient for a TE to drive several TRs at different locations simultaneously.

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